LEVEL II

(12)

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

WATERJET PROPULSOR PERFORMANCE PREDICTION IN PLANING CRAFT APPLICATIONS by Stephen B. Denny Allan R. Feller DTNSRDC SPD/0905-01

AD A075690

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by

Stephen B. Denny

and

Allan R. Feller

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DEPARTMENTAL REPORT

August 1979

DTNSRDC SPD/0905-01

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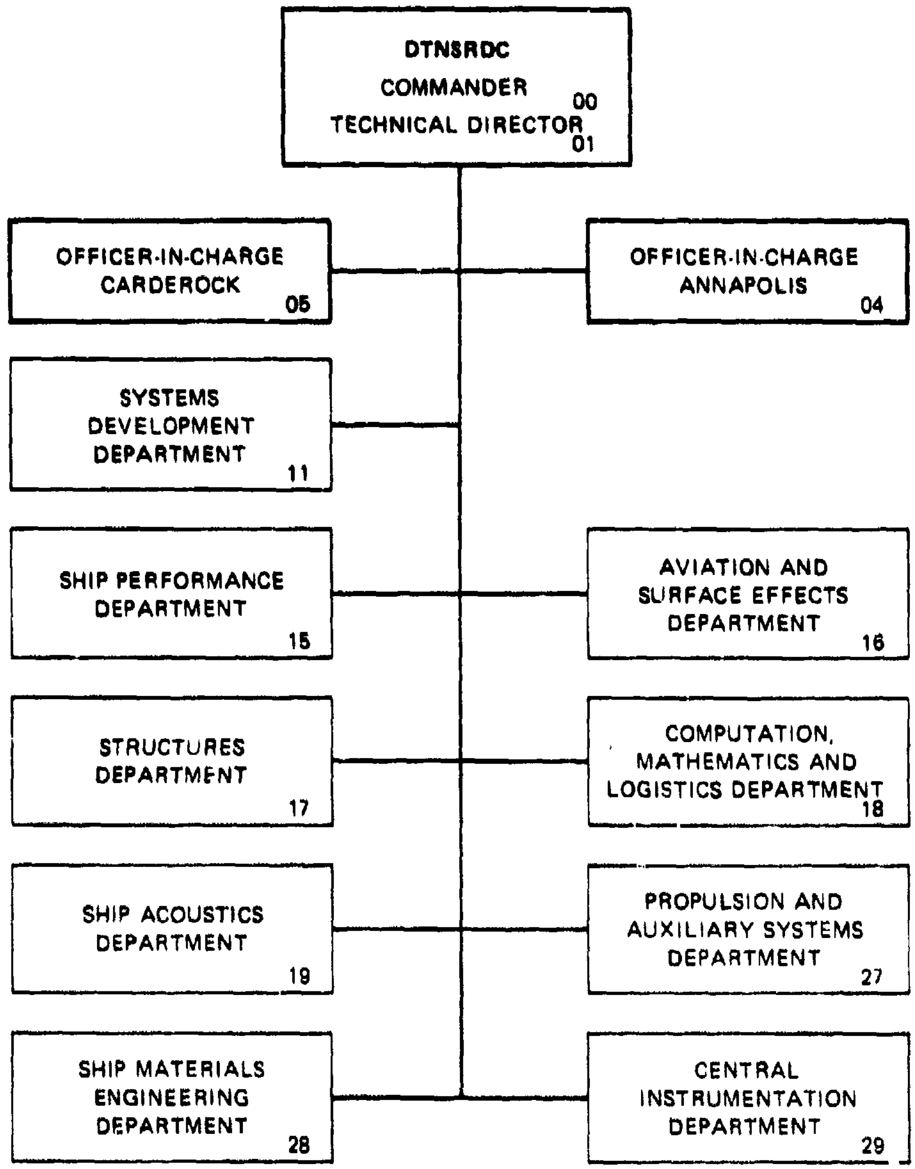

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER (14) DTNSRDC/SPD-0905-01	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) (6) WATERJET PROPULSOR PERFORMANCE PREDICTION IN PLANING CRAFT APPLICATIONS		5. TYPE OF REPORT & PERIOD COVERED (9) Final rept.
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) (10) Stephen B. Denny ~~and~~ Allan R. Feller		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS David W. Taylor Naval Ship R&D Center Bethesda, Maryland 20084		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Work Request N64281-78-WR-8-0080 Work Unit 1-1532-600
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Ship Engineering Center Norfolk Division Norfolk, Virginia 23511		12. REPORT DATE (11) August 1979
		13. NUMBER OF PAGES 32
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office) (12) 42		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Water-jet Propulsors
Planing Craft
Performance Prediction

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A performance prediction technique is presented for flush inlet waterjet systems installed on planing craft. The equations were derived based on empirical data and physical reasoning. This technique may be used to find an optimum waterjet configuration for a given planing craft, based on desired propulsive efficiency, weight, power, craft speed, waterjet size or any combination of these parameters. The useful domain of this technique is bound by the data base used in the derivation, which is a collection of commercially —

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S/N 0102-LF-014-6601

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(Block 20)
available waterjets. Radically new waterjet configurations would require an update of the equations based on empirical data.

Experimental and analytical work is recommended for the predictions of flush inlet head recovery and of pump cavitation characteristics. New data would improve the accuracy of the current mathematical model.

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NOTATION

Symbol	Description	Dimensions
A_I	Pump inlet area	L^2
$A_{i,i=0,3}$	Coefficients of waterjet weight equation	
A_j	Jet area	L^2
$B_{i,i=0,3}$	Coefficients of simplified weight equation	
$C_{i,i=1,2}$	Coefficients of specific speed equation	
D	Pump impeller diameter	L
D_h	Impeller hub diameter	L
E	Energy in jet	ML^2T^{-2}
e_j	Jet efficiency	1
e_m	Machinery efficiency	1
e_p	Pump efficiency	1
F	Coefficient of empirical power equation	
g	Acceleration due to gravity	LT^{-2}
H	Waterjet height	L
HS_{FS}	Total static pressure in the free stream, expressed as a column of water	L
HS_I	Total static pressure at the pump inlet, expressed as a column of water	L
H_{VP}	Vapor pressure of liquid, expressed as a column of water	L
h_L	Pressure losses, expressed as a column of water	L
IHR	Inlet head recovery	1

Symbol	Description	Dimensions
IVR	Inlet velocity ratio	1
J'	Effective advance coefficient	1
K	Inlet head recovery factor	1
L	Waterjet length	L
M	Power conversion coefficient	1
m	Mass of water flow through waterjet	M
N	Pump shaft revolution rate	T^{-1}
NPSH	Net positive suction pressure, expressed as a column of water	L
$P(P_{max})$	Waterjet input power (peak power)	ML^2T^{-3}
P_{jB}	Waterjet bollard output power	ML^2T^{-3}
Q	Waterjet volume flow rate	L^3T^{-1}
Q_B	Volume flow rate at bollard conditions	L^3T^{-1}
R	Craft resistance	MLT^{-2}
S_s	Suction specific speed	$L^{3/4}T^{-3/2}$
T	Waterjet thrust	MLT^{-2}
T_B	Thrust at bollard conditions	MLT^{-2}
T'	Underway thrust assuming no inlet head recovery	MLT^{-2}
V_h	Hump drag speed	LT^{-1}
V_I	Flow velocity into the pump inlet	LT^{-1}
V_j	Jet velocity	LT^{-1}
V_{jB}	Jet velocity at bollard conditions	LT^{-1}
V_s	Craft speed	LT^{-1}

Symbol	Description	Dimensions
W	Work	ML^2T^{-2}
W_w	Waterjet weight	MLT^{-2}
$\dot{W}$	Rate of work done	ML^2T^{-3}
x	Distance in direction of flow	L
ΔT	Differential thrust due to inlet head recovery	MLT^{-2}
ΔV_J	Differential jet velocity due to inlet head recovery	LT^{-1}
η_I	Inlet efficiency	1
ρ	Mass density of the working fluid	ML^{-3}
σ	Cavitation number based on pump inlet inflow velocity	1
σ_{TIP}	Cavitation number based on inflow velocity to impeller blade tip section	1

ABSTRACT

A performance prediction technique is presented for flush inlet waterjet systems installed on planing craft. The equations were derived based on empirical data and physical reasoning. This technique may be used to find an optimum waterjet configuration for a given planing craft, based on desired propulsive efficiency, weight, power, craft speed, waterjet size or any combination of these parameters. The useful domain of this technique is bound by the data base used in the derivation, which is a collection of commercially available waterjets. Radically new waterjet configurations would require an update of the equations based on empirical data.

Experimental and analytical work is recommended for the predictions of flush inlet head recovery and of pump cavitation characteristics. New data would improve the accuracy of the current mathematical model.

ADMINISTRATIVE INFORMATION

This work was funded by the Naval Ship Engineering Center, Norfolk Division (NAVSECNORDIV), under Work Request Number N64281-78-WR-8-0080, and carried out under DTNSRDC Work Unit 1-1532-600.

INTRODUCTION

Feasibility models have been developed to assist in the preliminary design of planing hulls.[1*] These models include weight, volume, and centers of gravity of the various components of the craft system and may be utilized to improve payload carrying ability by optimizing hull proportions. The inclusion of propulsor performance characteristics and specific fuel consumption data into the model allows the development of the most effective platforms for required missions. The use of high speed computers enables rapid successive iteration and model convergence to the most likely candidate preliminary craft design.

The existing feasibility model currently contains specific criteria relating to the sizing and performance of high speed propellers and

*References are listed on page 27.

required appendages. The purpose of this paper is the development of a formulation which is capable of estimating geometry and performance of flush inlet axial flow (or mixed flow) waterjets, and which may be included in the feasibility model.

Information related to geometry and full scale performance of flush inlet axial (or mixed flow) waterjets is in considerably shorter supply than comparable information for high speed propellers. Therefore, a secondary objective of this task is the organization of the performance prediction formulation in stages and with a degree of simplicity, so that it may be easily modified and updated as additional data become available.

ANALYSES

APPROACH

Available performance data for waterjets are usually presented by the manufacturer as curves showing output thrust versus craft speed at specific input horsepower and as plots of waterjet absorbed power versus pump revolution rate. Experiments conducted to produce such data are normally carried out on static test stands, where the pump inlet is connected to a supply pipe. Power input is determined from torque and RPM measurements on the impeller drive shaft and thrust is determined from mass flow. Therefore, flush inlet performance in underway conditions is not determined as part of the waterjet test programs and the inlet capabilities to efficiently deliver flow to the pump (with minimal induced drag and minimal internal losses) must be estimated.

In view of the character of available waterjet performance data, two methods of approach appeared to exist for developing formulations for waterjet performance prediction. One method would be similar to that of Kim[2] in which ducting losses are calculable based on duct geometry; and nozzle losses, inlet losses, and pump efficiencies are estimated. A second method would be to attempt to find a systematic relationship between existing waterjet performance data. Each method would require estimates to be made of flush inlet/diffuser performance.

The method involving calculation of head losses within the duct becomes quite involved when one considers the effects of internal shafting, stators, duct contractions and expansions, and frictional losses which are

dependent on surface roughness and Reynolds number. For this reason, the alternate method of attempting to "curve-fit" published waterjet data was chosen for the investigation.

DEFINING JET THRUST

To fully describe the procedure used, it is helpful to observe Figure 1. This figure shows a typically published waterjet thrust versus speed relationship (solid curve) at a given constant power and constant impeller revolution rate. This curve is developed from the general equation:

$$T = \rho Q(V_j - V_s) \tag{1}$$

where

T = thrust, pounds (N)
ρ = density of the fluid, lb sec^2/ft^4(kg/m^3)
Q = volume flow rate, ft^3/sec (m^3/s)
V_j = jet velocity, ft/sec (m/s)
V_s = craft speed, ft/sec (m/s)

At the zero craft speed (V_s=0), thrust is defined as $T_B = \rho Q_B V_{jB}$ where the subscript B defines bollard operating conditions.

If the assumption is made that volume flow rate (Q) does not change with craft speed, then

$$Q = Q_B$$

$$V_j = V_{jB}$$

and it follows from Equation (1) that the thrust versus speed relationship may be defined as:

$$T' = T_B - \rho Q_B V_s \tag{2}$$

Equation (2) has been plotted on Figure 1 as a dashed line. It is apparent from Figure 1 that at any given craft speed

$$T = T' + \Delta T \tag{3}$$

where ΔT is an additional thrust, resulting from inlet head recovery. The relationships shown in Figure 1 are based upon the initial constraints of constant power and constant revolution rate.

DETERMINATION OF JET AREA AND JET VELOCITY

With complete (including bollard) thrust versus speed performance prediction data, one can derive the specific bollard volume flow rate (Q_B), bollard jet velocity (V_{jB}), and jet area (A_j) in the following manner. Differentiating equation (1) with respect to V_s, we obtain:

$$\frac{dT}{dV_s} = \rho(V_j - V_s)\frac{dQ}{dV_s} + \rho Q\left[\frac{dV_j}{dV_s} - 1\right] \tag{4}$$

If we make the additional assumption that at or near $V_s = 0$, volume flow rate is constant ($Q = Q_B$), then

$$\left.\frac{dQ}{dV_s}\right|_{V_s \to 0} = 0 \quad \text{and} \quad \left.\frac{dV_j}{dV_s}\right|_{V_s \to 0} = 0$$

Therefore, at $V_s = 0$, Equation (4) becomes:

$$\frac{dT}{dV_s} = -\rho Q_B \, . \tag{5}$$

Since, by definition,

$$T_B = \rho Q_B V_{jB}$$

$$V_{jB} = \frac{T_B}{\rho Q_B} = \frac{-T_B}{(dT/dV_s)_{V_s = 0}} \tag{6}$$

and

$$A_j = Q_B/V_{jB} \tag{7}$$

The accuracy of this method for determining jet area and jet velocity is dictated by the accuracy with which one determines $(dT/dV_s)_{V_s = 0}$. In instances in which T versus V_s data are available for a number of power conditions, A_j values may be calculated for each power level and averaged to obtain the best estimate of A_j.

POWER ABSORPTION

In the manner described above, waterjet performance characteristics at bollard conditions, i.e., V_{jB}, Q_B, and A_j, can be determined. If the performance data under analysis relate to flush inlet axial or mixed flow waterjets alone, then the data undoubtedly have similar restrictions as follow:

1. They are derived from static test stand experiments in which supply pipes to the pump inlet offer as little flow restriction as possible.

2. Each bollard thrust value corresponding to a given power input exhibits the effects of inherent internal (frictional and shape related) losses in the pump and nozzle due to ducting, stators, shafting, impeller, etc.

3. In each case, the pump and nozzle centerlines lie as near as possible to the still water surface to allow the waterjet to exit into the air and still minimize the height over which the water is raised from the flush inlet to the pump inlet. This similarity suggests that at bollard conditions the total static pressure head at the pump inlet is approximately the same value for each set of waterjet data under consideration.

In view of these similarities, it appears that waterjet bollard performance data should yield good "power in-power out" estimates of waterjet unit efficiency from the pump inlet to the jet's vena contracta. Input power is measured in the test-stand experiments. Output power may be derived as follows.

Consider an element of water passing through the vena contracta. It has mass

$$dm = \rho A_j dx$$

where

dm = mass of element

dx = thickness of element in direction of flow

At the vena contracta, static pressure in the jet is ambient pressure. With no further pressure change, the energy of the element is solely kinetic and it may be evaluated as

$$dE = 1/2 \; dmV_{jB}^2$$

$$dE = 1/2 \;\; \rho A_j dxV_{jB}^2$$

where dE is the kinetic energy of the element. The power in the jet is the time rate of change of energy, and since ρ, A_j, and V_{jB} are all time invariant (steady-state operation)

$$\frac{dE}{dt} = P_{jB} = \frac{1}{2}\rho A_j \frac{dx}{dt} V_{jB}^2$$

where P_{jB} = power in jet of water at bollard condition. This expression is made simpler by use of the relations

$$\frac{dx}{dt} = V_{jB}$$

$$Q_B = A_j V_{jB}$$

$$T_B = \rho Q_B V_{jB}$$

Substituting,

$$P_{jB} = \frac{1}{2}\rho A_j V_{jB} V_{jB}^2$$

$$P_{jB} = \frac{1}{2}\rho Q_B V_{jB}^2$$

$$P_{jB} = \frac{1}{2} T_B V_{jB}$$

The last expression defines the output power of the waterjet system at the bollard condition (P_{jB}).

Since thrust times velocity is proportional to power, a plot was generated in which the product of bollard thrust and bollard jet velocity was plotted versus input horsepower for a number of waterjets. In spite of the wide range of horsepower values and the number of different waterjets under consideration, the plotted data were quite consistent. Plotted on 3 cycle by 3 cycle log paper to compress the ordinate and abscissa axes, the data in Figure 2 tend to represent the functional relationship

$$T_B V_{jB} = FP^{1.0556} \tag{8}$$

where

F = 620.517 for British units of lbs, ft/sec, hp
F = 1146.77 for SI units of N, m/s, kW
T_B = bollard thrust in lbs (N)
V_{jB} = bollard jet velocity in ft/sec (m/s)
P = input power in hp (kW)

Equation (8) can be quite useful for waterjet bollard performance prediction and it can be extended to provide estimates of available water-jet thrust underway. If constant volume flow-rate and jet velocity (at constant impeller RPM) are assumed over the speed range of the craft, then equation (2) may be used:

$$T' = T_B - \rho Q_B V_s$$

Multiplying through by V_{jB}, this becomes

$$T'V_{jB} = T_B V_{jB} - \rho Q_B V_{jB} V_s$$

or, since $\rho Q_B V_{jB} = T_B$,

$$T'V_{jB} = T_B V_{jB}\ (1-V_s/V_{jB}) \tag{9}$$

From Equation (8)

$$T_B V_{jB} \simeq FP^{1.0556} = f_1(P)$$

therefore, for no inlet head recovery (ram effect) or loss, the underway thrust of a waterjet system may be approximated as

$$T' = \frac{f_1(P)}{V_{jB}}\ (1-V_s/V_{jB}) \tag{10}$$

INTERNAL SYSTEM LOSSES

According to Kim[2], jet efficiency (e_j) is defined as

$$e_j = \frac{\dot{W}}{e_m e_p \mathrm{BHP}} = \frac{2(V_j/V_s-1)}{\left(\frac{V_j}{V_s}\right)^2 - 1 + \frac{2gh_L}{V_s^2}}$$

where

$\dot{W}$ = rate of work done (power), hp (kW)

e_m = machinery efficiency

e_p = pump efficiency

BHP = brake horsepower (kW)

$\frac{2gh_L}{V_s^2}$ = nondimensional losses

Since $e_m \cdot$ BHP is equal to input power (P), then

$$e_j = \frac{\dot{W}}{e_p P} = \frac{TV_s}{Me_p P}$$

where

M = 550 for British units of hp, ft/sec, lb

M = 1000 for SI units of kW, m/s, N

If we again assume no inlet head recovery (or loss), then

$$T = T'$$

$$V_j = V_{jB}$$

$$e_j e_p = \frac{T' V_s}{MP} = \frac{f_1(P)}{V_{jB}} \left(1 - \frac{V_s}{V_{jB}}\right) \frac{V_s}{MP}$$

and

$$2e_p \frac{(V_{jB}/V_s) - 1}{\left(\frac{V_{jB}}{V_s}\right)^2 - 1 + \frac{2gh_L}{V_s^2}} = \frac{F}{M} (P^{.0556}) \left(\frac{V_s}{V_{jB}}\right)\left(1 - \frac{V_s}{V_{jB}}\right)$$

or

$$2e_p \frac{V_{jB}^2}{V_s^2 \frac{F}{M}(P^{.0556})} = \left(\frac{V_{jB}}{V_s}\right)^2 -1 + \frac{2gh_L}{V_s^2}$$

If pump efficiency (e_p) is assumed to be 0.90, then

$$\frac{1.8M}{FP^{.0556}} \left(\frac{V_{jB}}{V_s}\right)^2 = \left(\frac{V_{jB}}{V_s}\right)^2 -1 + \frac{2gh_L}{V_s^2}$$

which leads to a fairly simple relationship for nondimensional internal losses as a function of input power and bollard jet velocity ratio

$$\frac{2gh_L}{V_s^2} = \left(\frac{V_{jB}}{V_s}\right)^2 \left(\frac{1.8M}{FP^{.0556}} -1\right) + 1 \qquad (11)$$

Table 1 shows representative calculated values of $2gh_L/V_s^2$ for ranges of input power and bollard jet velocity ratio.

Table 1

Calculated Nondimensional Head Losses as Functions of Input Power and Jet Velocity Ratio (Pump Efficiency = 0.90)

Input Power		V_{jB}/V_s =			
Horsepower	(Kilowatts)	1	2	3	4
100	75	1.24	1.94	3.12	4.77
500	373	1.13	1.52	2.17	3.08
1,000	746	1.09	1.35	1.78	2.39
2,000	1491	1.05	1.18	1.41	1.73
5,000	3729	0.99	0.98	0.95	0.90
10,000	7457	0.96	0.83	0.61	0.30

Figure 3 represents a plot of the values in Table 1. Superimposed on Figure 3 are curves which correspond to $e_j \cdot e_p = 0.4$ and $e_j \cdot e_p = 0.5$ (assuming $e_p = 0.9$). These data show that without significant inlet head recovery (which would increase the thrust produced for a given power), it is unlikely that propulsive efficiency ($e_j \cdot e_p$) will exceed 50%. These data also show that over a wide range of power, peak propulsive efficiency will be obtained at jet velocity ratios (V_{jB}/V_s) near 2.0. This does not mean that the optimum jet velocity ratio for a given operating condition will not be significantly different from 2.0 when total system weight and size are taken into account.

Equation (11) was derived assuming pump efficiency equal to 0.90. The formulation for head losses will be quite sensitive to the value of pump efficiency assumed. Therefore, Figure 3 is intended only to exhibit trends and indicate typical head loss values as functions of jet velocity ratios.

INLET EFFICIENCY

With the importance of inlet head recovery in overall waterjet efficiency shown above, a closer look at inlet efficiency was deemed necessary. As described earlier, available waterjet performance prediction data are generally derived from static test stand data obtained with the pump connected to a supply pipe and reservoir. The pump performance is characterized over ranges of shaft rpm, power, and static pressure at the pump inlet (varied by changing the reservoir static water level height relative to the pump height). With this information, pump mass flows for various rpm and power conditions can be related to the net positive suction head (NPSH) at the pump inlet, defined as

$$\mathrm{NPSH} = V_I^2/2g + H_{S_I} - H_{VP} \qquad (12)$$

where

V_I = pump inlet velocity, ft/sec (m/s)

H_{S_I} = total static head at pump inlet, ft (m) of H_2O

H_{VP} = vapor pressure of liquid, ft (m) of H_2O

g = acceleration due to gravity, ft/sec^2 (m/s^2)

For the underway performance prediction of a flush inlet waterjet system, an inlet head recovery (IHR) is estimated which yields the total head (NPSH) at the pump inlet. Pump thrust is then obtained from applicable test and calibration data and usable thrust is derived after momentum loss due to craft speed is subtracted. In this procedure, the most critical item is the inlet head recovery. If inlet efficiency (η_I) is defined as the ratio of total head delivered to the pump inlet to the total head available, then

$$\eta_I = \frac{NPSH}{\frac{V_{s}^2}{2g} + H_{S_{FS}} - H_{VP}} \qquad (13)$$

where $H_{S_{FS}}$ is the total static head in the free stream, in ft (m) of H_2O. Inlet efficiency is quite sensitive to inlet velocity ratio (IVR = V_I/V_s) and inlet geometry.

Since there existed obvious differences in the inlet head recoveries and inlet efficiencies claimed by various manufacturers, and since inlet head recovery is difficult to determine without a sophisticated test program (conducted with the specified flush inlet/diffuser geometry), it was decided that inlet head recovery would be an input variable in the waterjet performance formulation. Specifically, the user of the procedure for predicting waterjet performance would choose a factor of safety and thereby establish the value of inlet head recovery (IHR) (between zero and the most optimistic values), to be assumed during a given overall performance prediction. Should more precise values of IHR be available for a given inlet/diffuser geometry over appropriate ranges of inlet velocity ratios (IVR), then the procedure should use the more exact data.

JET VELOCITY INCREASE

Since T, V_s, V_{jB}, and A_j are ultimately available from performance

data, the increased jet velocity ($\Delta V_j \equiv V_j - V_{jB}$) due to inlet head recovery may be determined at any given operating condition. Figure 4 represents $\Delta V_j/V_s$ data derived for 20 and 30 knot speed conditions. The individual data points were derived from performance prediction data of many different manufacturers in the following manner:

$$T = \rho Q(V_j - V_s) = \rho V_j A_j (V_j - V_s)$$

$$T = \rho (V_{jB} + \Delta V_j) A_j (V_{jB} + \Delta V_j - V_s)$$

These points have been fitted with a mathematically defined curve as shown. Use of the curve fit equation for $\Delta V_j/V_s$ may proceed as follows:
For $0 \leq K \leq 1$,

$$\frac{\Delta V_j}{V_s} = K \frac{1}{\left(\frac{V_{jB}}{V_s} + 1\right)^{1.737}} \tag{14}$$

where K is an inlet head recovery factor. The V_j values calculated may be consistent with available data (K = 1) or pessimistic (K = 0), assuming no increased jet velocity due to inlet head recovery. Values of ΔV_j may also be calculated which correspond to any K value between 0 and 1, depending on the factor of safety desired.

CAVITATION PERFORMANCE

One of the more difficult areas in waterjet performance prediction is that of performance breakdown due to cavitation. In high speed applications, cavitation may occur in the inlet/diffuser, restricting inflow and resulting in sudden deterioration of thruster performance. In general, the prediction of inlet cavitation is a very elusive task, and each geometry and application must be handled independently.

Somewhat more approachable are the criteria dealing with pump cavitation and pump performance breakdown. One of these criteria is "suction specific speed," which is defined as

$$S_s = \frac{C_1 N\sqrt{Q}}{(NPSH)^{3/4}}$$

where

C_1 = 1 for British units of rev/min, gallons/min, ft (H_2O)

C_1 = 3098.9 for SI units of rev/s, m^3/s, m (H_2O)

N = shaft revolution rate, rev/min (rev/s)

Q = volume flow rate, gallons/min (m^3/s)

NPSH = net positive suction head, ft (m) of H_2O

Other than offering a rather confusing combination of units, suction specific speed represents a peak numerical value corresponding to a cavitation limit beyond which a given pump or geometrically similar scaled version of the pump cannot operate. Typically, S_s ratings for axial flow and mixed flow pumps range between 16,000 and 20,000.

To further evaluate the meaning and significance of suction specific speed, an analysis of the formulation was carried out. For convenience

$$S_s = \frac{C_2 N\sqrt{Q}}{(NPSH)^{3/4}}$$

where

C_2 = 1271.14 for English units of rps, ft^3/s, ft (H_2O)

C_2 = 3098.9 for SI units of rps, m^3/s, m (H_2O)

N = shaft revolution rate (rps)

Q = volume flow rate, ft^3/s (m^3/s)

NPSH = net positive suction head, ft (m) of H_2O

or

$$S_s = \frac{C_2 N\sqrt{V_I A_I}}{(V_I^2/2g + H_{S_I} - H_{VP})^{3/4}}$$

where

V_I = average flow velocity into the pump inlet, ft/s (m/s)

A_I = open area of the pump inlet, ft^2 (m^2)

If we assume that the open area of the pump inlet is approximately equal to the open area (between hub and casing) at the pump impeller, then

$$A_I = \pi/4\ D^2(1-D_h^2/D^2)$$

where

D_h = hub diameter, ft (m)

D = impeller diameter, ft (m)

Multiplying both numerator and denominator by $(2g)^{3/4}$ suction specific speed becomes

$$S_s = \frac{C_2(2g)^{3/4} N\sqrt{V_I} D\sqrt{\pi/4}\ \sqrt{1-D_h^2/D^2}}{(V_I)^{3/2}(1+\sigma)^{3/4}}$$

where σ is the cavitation number based on inflow velocity, defined as

$$\sigma = \frac{2g(H_{S_I} - H_{VP})}{V_I^2}$$

Therefore,

$$S_s = \frac{1.49045\ C_2(g)^{3/4}\ \sqrt{1-D_h^2/D^2}}{J'(1+\sigma)^{3/4}}$$

where J' is the effective advance coefficient, defined as $J' = V_I/ND$.

If S_s is identified as a function of impeller tip blade section cavitation number, then

$$S_s = \frac{1.49045\ C_2(g)^{3/4}\ \sqrt{1 + (D_h/D)^2}}{J'[1+\sigma_{TIP}(1+\ \pi/J')^2]^{3/4}} \tag{15}$$

Figure 5 has been generated from Equation (15) assuming typical ranges of σ_{TIP}, J', and a hub-diameter ratio (D_h/D) of 0.5. It is apparent from the data that the suction specific speed range from 16,000-20,000 corresponds to impeller tip cavitation numbers (σ_{TIP}) between 0.07 and 0.05 at optimum operating conditions (considering cavitation performance). This is realistic, since, in conventional propeller theory, local cavitation numbers from 0.10 to 0.05 normally define the range from cavitation inception to significant cavitation formation, respectively on blade sections. Also, the peak values of suction specific speed (S_s) for constant values of impeller tip cavitation number (σ_{TIP}) lie between effective advance coefficients (J') of 0.4 and 0.6 This range of effective advance coefficient was common for the specific pumps under consideration in this program. As a consequence of these data analyses, it would appear that axial and mixed flow waterjet cavitation performance prediction methods can predict the maximum shaft revolution rates practical to avoid cavitation. It would appear that regardless of power input, maximum shaft revolution rates which yield σ_{TIP} values near 0.05 and effective advance ratios between 0.4 and 0.5 are practical. Should waterjet systems fail to meet these criteria, it may be the result of separation and/or cavitation in the inlet/diffuser stage; it may also reflect the use of a lower level of technology for the system design.

WATERJET WEIGHT AND VOLUME

Weight information on waterjets was quite limited and in only a few cases was it complete for any one waterjet configuration. Therefore, estimates were made, where necessary, to generate tabulated weights (wetted) and dimensions of a consistent series of waterjets. The total

weight values derived included:

1. Wet weight of total unit forward of craft transom (reversing gear not included)

2. Materials for sea water environment

3. Reasonably hydrodynamically smooth inlet/diffuser and inlet transition (when not already included in specification data)

It should be realized that higher power waterjets available today reflect a considerable financial investment in their technological development. Lower power units, although adequate performers, may not have had an equivalent technology investment. Therefore, the trends of waterjet weight to peak input power appear to range from near 2.0 lb/hp (12 N/kW) at lower power ratings to near 1.0 lb/hp (6.0 N/kW) at higher power ratings.

For all available pointwise weight data, a "least squares" curve fit of a third order polynomial produced reasonable agreement.

The equation produced was of the form:

$$WT(V_{jB})^3 = A_0 + A_1 P_{max} + A_2 P_{max}^2 + A_3 P_{max}^3$$

where

WT = wetted weight, lbs (N)

V_{JB} = bollard jet velocity, ft/s (m/s)

P_{max} = peak power, hp (kW)

Since weight is determined as a function of peak power and V_{jB} is related to power by:

$$T_B V_{jB} = \rho A_j V_{jB}^3 = FP^{1.0556}$$

where ρ = density of water, lb sec^2/ft^4 (kg/m^3)

then

$$WT = \rho A_j (B_0 P_{max}^{-1.0556} + B_1 P_{max}^{-0.0556} + B_2 P_{max}^{0.0444} + B_3 P^{1.9444})$$

where the coefficients B_o, B_1, B_2, and B_3 are:

	English	SI
B_0 =	-695241	-47385.8
B_1 =	+4321.3	+394.969
B_2 =	+1.2156	+.14900
B_3 =	-0.0000395	$-.649 \times 10^{-5}$

These coefficients were generated on the basis of data ranging from 250 to 15,000 peak horsepower (186 to 11186 kW). The curve produced reasonable agreement with the data over this range, but any extrapolation to higher or lower power levels would be questionable.

To determine the compartment size necessary to house a waterjet with inlet diffuser and inlet transition fairing, the following relationships are used:

waterjet width, $W_w = 1.10\ D$
waterjet length, $L = W_w/0.23$
waterjet height, $H = 0.37\ L$

where

D = impeller diameter, ft (m)

For sizing a watertight compartment, add 1.5 feet (0.46 m) above the calculated waterjet height and 3.0 feet (0.91 m) to the calculated waterjet width. These additional clearances allow space for inspection and maintenance.

EXAMPLE USING THE WATERJET PERFORMANCE PREDICTION METHOD FOR PLANING CRAFT

The following is a sample problem concerned with flush inlet waterjet propulsion and the details of its solution using the procedure described in this report.

Problem: A planing craft equipped with port and starboard engines rated at 400 horsepower (298 kW) each will be fitted with flush inlet waterjets. At design displacement, the resistance versus speed predictions are as shown below with the hump drag speed (V_h) occurring at 12 knots. Predict the maximum speed capability, assuming a 10% thrust margin is desired at the hump drag speed, and a waterjet unit weight limit of 700 pounds (3114 N) maximum. Determine the approximate compartment size required for each waterjet unit.

Craft Speed V_s (Knots)	9	12	15	18	21	24	27	30
Resistance R (lb)	4000	4600	4200	4148	4382	4652	4958	5300
Resistance R (N)	17792	20462	18683	18451	19492	20693	22054	23576

$$WT = \rho A_j (B_0 P^{-1.0556} + B_1 P^{-0.0556} + B_2 P^{0.9444} + B_3 P^{1.9444})$$

where the constants B_0, B_1, B_2, and B_3 are:

	English	SI
B_0 =	-695241	-47385.8
B_1 =	+4321.3	+394.969
B_2 =	+1.2156	+0.14900
B_3 =	-0.0000395	-0.649×10^{-5}

For P_{max} = 400 hp (298 kW), the equation becomes

$$WT\ (lb) = \rho A_j\ (2194)$$

$$WT\ (N) = \rho A_j\ (204.3)$$

Therefore,

$$(A_j)_{max} = \frac{700}{(1.9905)(2194)} = 0.16\ ft^2$$

$$= \frac{3114}{(1025.86)(204.3)} = 0.01486\ m^2$$

Each waterjet will absorb 400 horsepower (298 kW) therefore

$$T_B V_{jB} = \rho A_j V_{jB}^3 = 620.517\ (400)^{1.0556}\ ftlb/sec$$

$$= 1146.77\ (298)^{1.0556}\ Nm/s$$

$$\rho A_j V_{jB}^3 = 346328\ ftlb/sec$$

$$= 469093\ Nm/s$$

and bollard jet velocity (V_{jB}) can be determined for a number of different jet areas.

CONFIGURATION

		A	B	C	D	Units
A_j	=	0.10	0.12	0.14	0.16	ft^2
	=	(0.00929	(0.01115)	(0.01301)	(0.01486	(m^2)
V_{jB}	=	120.30	113.20	107.50	102.80	ft/sec
	=	(36.67)	(34.50)	(32.77)	(31.33)	(m/s)
Q_B	=	12.03	13.58	15.05	16.45	ft^3/sec
	=	(0.3407)	(0.3845)	(0.4262)	(0.4658)	(m^3/s)
T_B	=	2881	3060	3220	3366	lb
	=	(12815)	(13611)	(14323)	(14972)	(N)
V_{jB}/V_h	=	5.94	5.59	5.31	5.08	--
$\Delta V_j/V_h$	=	0.035	0.038	0.041	0.043	--
ΔV_j	=	0.71	0.77	0.83	0.87	ft/sec
	=	(0.22)	(0.23)	(0.25)	(0.27)	(m/s)
V_j	=	121.01	113.97	108.33	103.67	ft/sec
	=	(36.884)	(34.738)	(33.019)	(31.599)	(m/s)
T	=	2427	2551	2659	2754	lb
(at 12 knots)	=	(10795)	(11347)	(11827)	(12250)	(N)

Since craft resistance at 12 knots (V_h) is 4600 lbs (20461 N) and a 10% thrust margin is required, then the necessary thrust per waterjet at 12 knots is

$$\frac{4600 + 460}{2} = \frac{5060}{2} = 2530 \text{ lb } (11253 \text{ N})$$

From the calculations above, it appears that configuration B produces a thrust at 12 knots which is sufficiently near the required 2530 lbs (11253 N). The selection of a configuration which produces more than the required thrust at 12 knots will lead to a reduced top speed capability due to increased momentum losses. Therefore, we can choose configuration B as the most appropriate unit to satisfy hump speed thrust and maximum speed requirements.

To determine the maximum speed capability of the subject planing craft equipped with two water jet units of the configuration B type, the following calculations are carried out.

								Units
V_s	=	15	18	21	24	27	30	Knots
V_{jB}	=	113.2 (34.50)	113.2 (34.50)	113.2 (34.50)	113.2 (34.50)	113.2 (34.50)	113.2 (34.50)	ft/sec (m/s)
V_{jB}/V_s	=	4.47	3.73	3.19	2.79	2.48	2.24	--
$\Delta V_j/V_s$	=	0.052	0.067	0.083	0.099	0.114	0.130	--
ΔV_j	= =	1.32 (0.402)	2.05 (0.625)	2.94 (0.896)	4.00 (1.219)	5.21 (1.588)	6.59 (2.009)	ft/sec (m/s)
V_j	=	114.5 (34.90)	115.2 (35.11)	116.1 (35.39)	117.2 (35.72)	118.4 (36.09)	119.8 (36.52)	ft/sec (m/s)
ρQ	=	27.35 (399.0)	27.52 (401.5)	27.73 (404.6)	27.99 (408.4)	28.28 (412.6)	28.62 (417.6)	slug/sec (kg/s)
$V_j - V_s$	=	89.18 (27.18)	84.82 (25.85)	80.66 (24.59)	76.69 (23.38)	72.83 (22.20)	69.17 (21.08	ft/sec (m/s)
2T	= =	4878 (21698)	4668 (20764)	4473 (19897)	4293 (19096)	4119 (18322)	3959 (17611)	lb (N)
R	=	4200 (18683)	4148 (18451)	4382 (19492)	4652 (20693)	4958 (22054)	5300 (23576	lb (N)

Assuming that thrust deduction is negligible, a plot of thrust capability versus craft speed superimposed on a plot of craft resistance versus speed yields the predicted maximum speed of the craft (21.5 knots).

The next step is to determine the size of the waterjet (configuration B). The waterjet unit weight limit and maximum power were prescribed. These criteria led to the selection of a relatively small range of jet areas. The selection of the pump diameter (impeller diameter) and consequent waterjet dimensions will be governed by the requirement that the pump be as small as possible and still operate relatively cavitation free. Such a requirement will insure that the waterjet size is realistic, since the waterjet equation was originally developed from data for existing waterjets with known cavitation limits.

Figure 5 in the text was generated assuming an impeller hub to tip diameter ratio (D_h) of 0.5. If it is assumed that the pump will be

essentially free of cavitation problems if the impeller tip cavitation number (σ_{TIP}) is greater than or equal to 0.06, then a suction specific speed (S_s) of 17600 should be achievable if the pump operates at an effective advance coefficient (J') of 0.50. This value of suction specific speed appears quite conservative in view of current technology. To proceed, the value of inlet velocity ratio must be assumed at which the inlet is most efficient and at which the inlet head recovery allows the generation of V_j/V_s values assumed in the previous calculations. For this example, it is assumed that the maximum inlet efficiency occurs at an inlet velocity ratio (V_I/V_s) of 0.80. Therefore, at V_s = 21.5 knots, V_I = 29.03 ft/sec (8.848 m/s). Since

$$Q = V_j A_j = A_I V_I$$

$$A_j = 0.12 \text{ ft}^2 \ (0.01115 \text{ m}^2)$$

and

$$V_j = V_{jB} + \Delta V_j = 113.2 + 3.1 = 116.3 \text{ ft/sec}$$

$$= (34.50 + 0.94 = 35.44 \text{ m/s})$$

then

$$A_I = A_j \frac{V_j}{V_I}$$

$$A_I = 0.12 \frac{116.3}{29.03} = 0.48 \text{ ft}^2$$

$$= (0.01115 \frac{35.44}{8.848} = 0.045 \text{ m}^2)$$

Knowing the impeller area (A_I) and with hub to tip diameter ratio (D_h/D) of 0.5, the pump diameter may be found:

$$A_I = \frac{\pi}{4} D^2 \left(1 - \frac{D_h^2}{D^2}\right) = \frac{\pi}{4} D^2 \ (0.75)$$

$$D^2 = \frac{16A_1}{3\pi} = 0.81\ \text{ft}^2\ (0.075\ \text{m}^2)$$

$$D = 0.9\ \text{ft}\ (0.27\ \text{m})$$

From the simple relationships given in the main portion of this report,

waterjet width $W_w = 1.10\ (D) = 1.10\ (0.9) = 0.99$ ft (0.30 m)

waterjet length $L = W_w/0.23 = 4.3$ ft (1.3 m)

waterjet height $H = 0.37\ L = 1.6$ ft (0.49 m)

The compartment desired to house each unit (allowing 1.5 ft (0.46 m) above and on each side for maintenance and inspection) will be 4.3 feet (1.3 m) long, 3.1 feet (0.95 m) high, and 3.99 feet (1.22 m) wide.

CONCLUSIONS AND RECOMMENDATIONS

The mathematical model and associated formulations presented herein comprise the basis of a fairly simple but workable waterjet propulsor performance prediction method. In using the method, one must remain aware of the assumptions which have been imposed during the method's development and evaluate the merit of each on the basis of the specific application problem. The assumptions are restated below.

The current mathematical model considers only flush inlet waterjets. Inlet drag and thrust deduction are considered negligible. Nominal wake fraction, per se, is not considered; however, differential thrust due to inlet head recovery is included in the procedure.

Jet cross-sectional areas which are determined from bollard thrust data and the slope of thrust versus speed curves (at $V_s = 0$) refer to the actual jet area at the "vena contracta" and are not intended to approximate specific nozzle areas.

In the development of the formulations, it has been assumed that head recovery for a flush inlet waterjet is equal to zero at the bollard

condition. In reality, depending on the inlet geometry, there may be significant inlet head loss at zero speed and the inlet conditions could become more favorable at slight ahead speeds.

The relationship $T_B V_{jB} = FP^{1.0556}$ was developed by curve-fitting waterjet bollard performance data for a number of waterjet pumps. It was assumed that in each pump test, the power losses were due to effects internal to the pump/nozzle system. These effects are frictional and shape related losses in the pump and nozzle due to ducting, stators, shafting, and the impeller.

In predicting waterjet performance from pump test data, one must assume the height (and consequent head loss) that the water must be raised from the flush inlet to the pump inlet. Normally this height is kept to the minimum possible within the hardware constraints of the pump and craft design.

Calculated waterjet losses have been presented in this paper showing the effects of jet velocity ratio and horsepower. As stated earlier, these calculations assumed a pump efficiency of 0.9. Should losses be calculated assuming other values for pump efficiency, the changes in calculated values would be significant.

A relationship has been presented which allows an incremented jet velocity increase to be determined as a function of bollard jet velocity and craft speed. This development assumes that presently "claimed" inlet head recovery values are achievable. For a specific flush inlet geometry, however, this may or may not be the case. The designer may choose any margin of safety he desires by designating a finite value of the "K" factor (where $0 \leq K \leq 1$).

In the cavitation analysis which presented suction specific speed (S_s) as a function of impeller tip cavitation number (σ_{TIP}), effective impeller advance coefficient (J'), and impeller hub to tip diameter ratio (D_h/D), the pump inlet area was assumed to be equal to the open area between the impeller hub and the pump casing. Also the generation of data for Figure 5 assumed an impeller hub to tip diameter ratio of 0.5. A change in either of these assumptions would alter the functional elationship between S_s, σ_{TIP}, and J'.

In order to develop a relationship for the wet weight of an installed waterjet propulsion unit, it was necessary in some cases to extrapolate weights and linear dimensions to account for flush inlet flow transition hardware (ahead) of the pump inlet. In all cases, those weight and dimension increases were based on those increases observed for the few existing installations. No attempt was made to estimate the weight of reversing and steering gear or the control hardware mounted aft of a craft transom.

A sample problem and the details of its solution are presented in the body of this paper. It is recommended that the procedure developed here be programmed for a computer and fully exercised to determine its degree of flexibility.

Further experimental and analytical work is recommended in the area of flush inlet head recovery. Inlet head recovery appears to be a sensitive function of inlet velocity ratio. At higher speed conditions, the thrust due to inlet head recovery (or loss) may be significant, appreciably affecting maximum speed capability.

Additional work is needed in establishing pump cavitation criteria. Firm guidelines should exist which define suction specific speed (or similar factors for a given pump configuration) in a consistent manner. Specifically, cavitation limits should define either peak volume flow capability, the initial point of performance breakdown due to cavitation, or the maximum, cavitation erosion-free, operating point. Claims of very high suction specific speed capabilities are of little value if pump operation at those conditions leads to excessive material erosion and premature failure. These cavitation limits should also be defined for the entire geometric range of pumps, from propeller-types to centrifugals, and including axial-flow and mixed-flow waterjet pumps.

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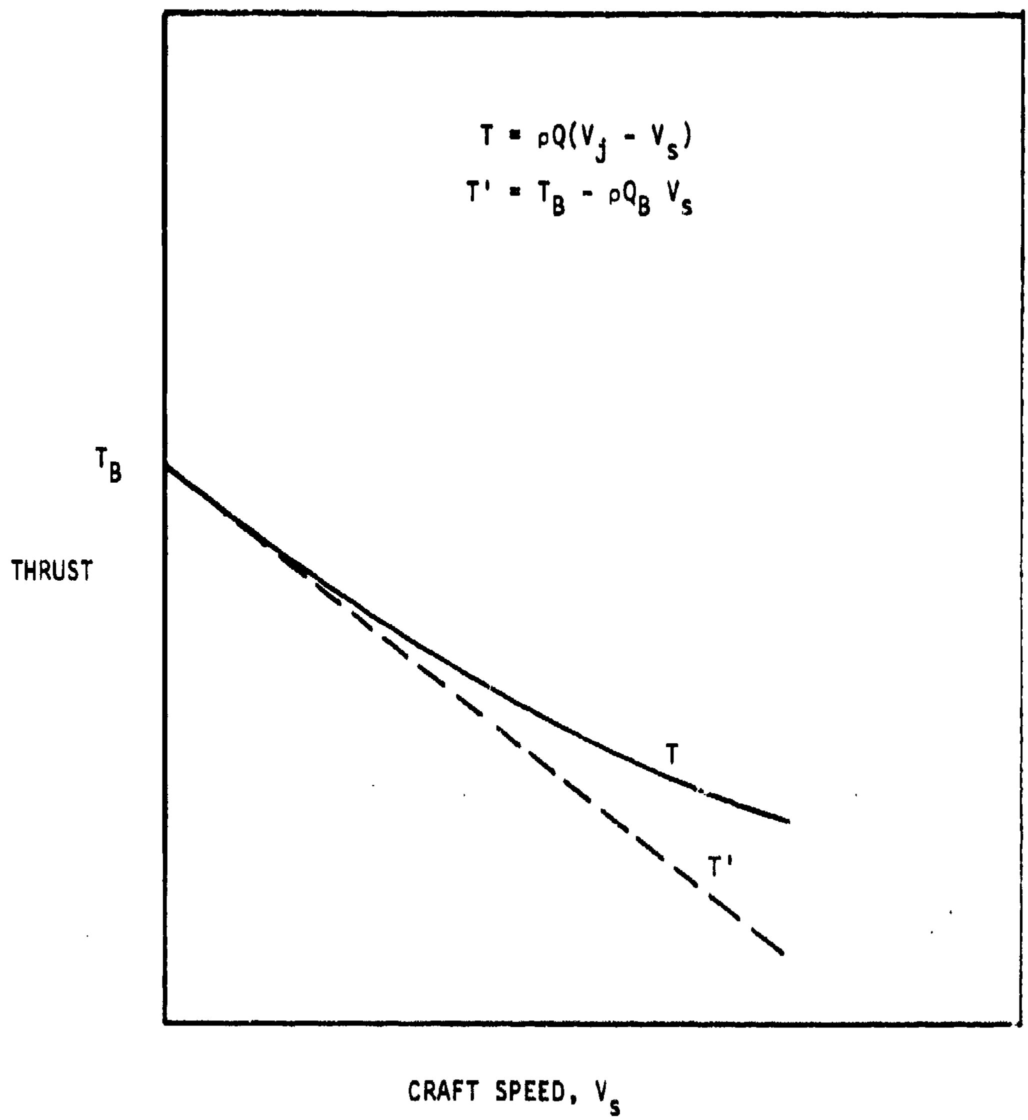

Figure 1 - Typical Waterjet Performance Prediction at Constant Input Horsepower

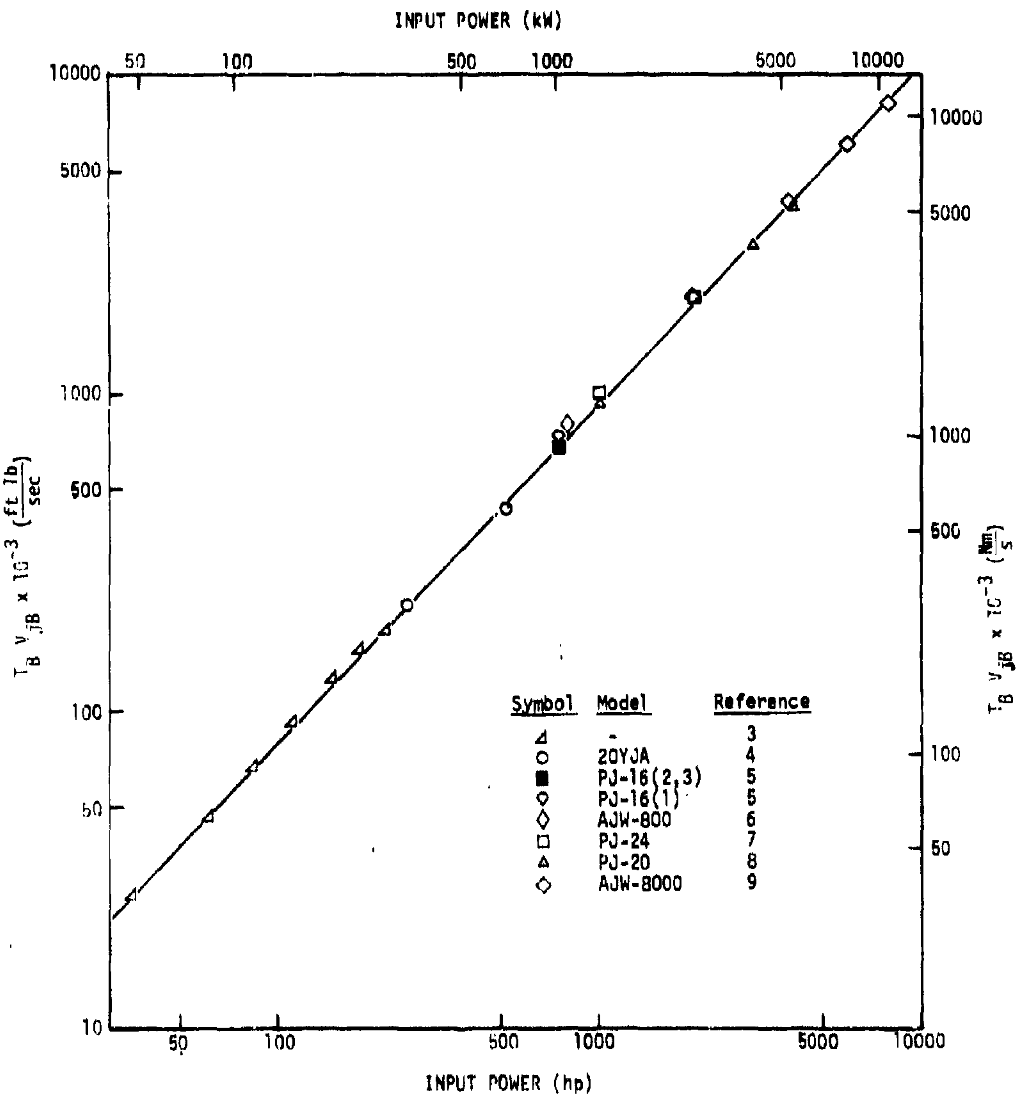

Figure 2 - Relationship Between Input Power and Bollard Performance

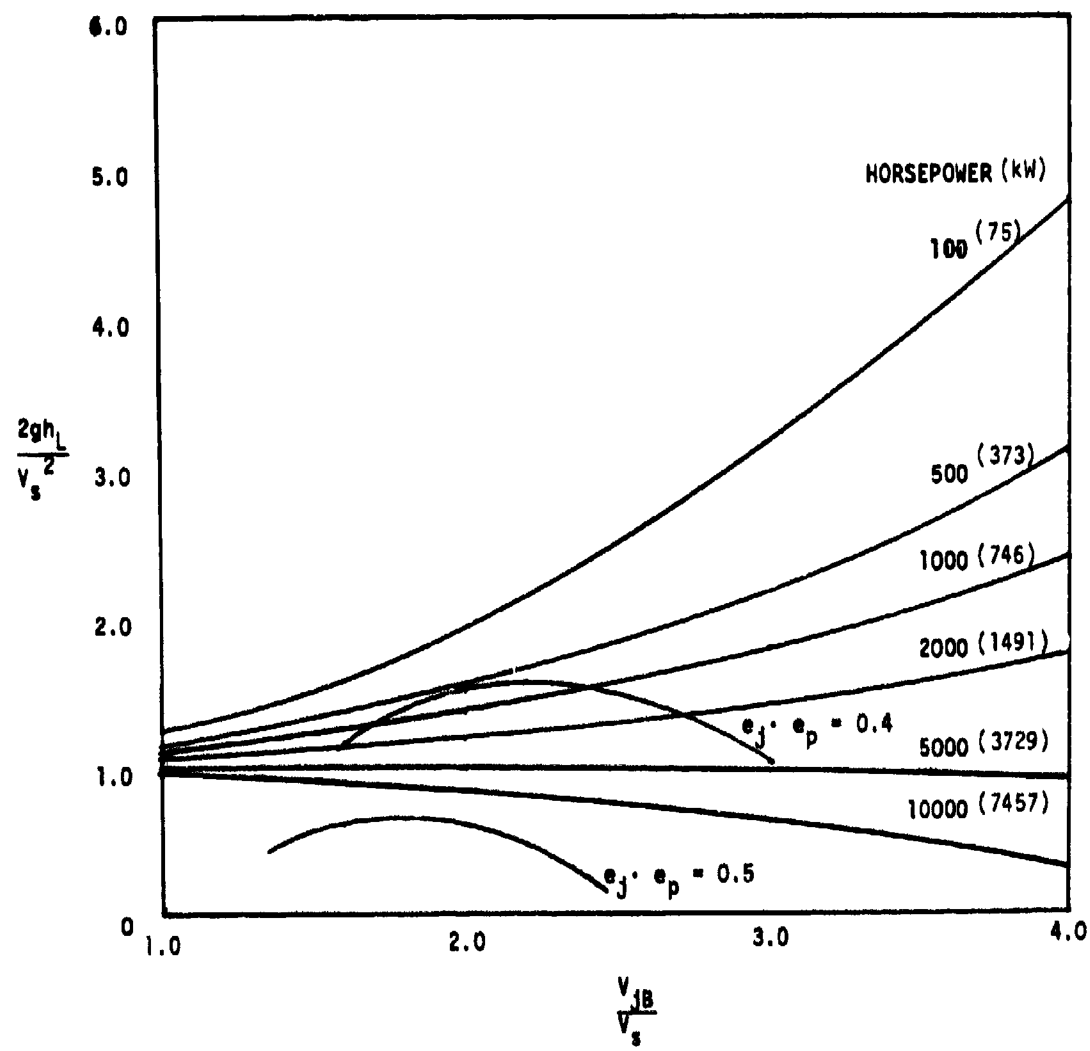

Figure 3 - Nondimensional Head Losses versus Bollard Jet Velocity Ratio (V_{jB}/V_s) for a Range of Horsepowers ($e_p = 0.9$)

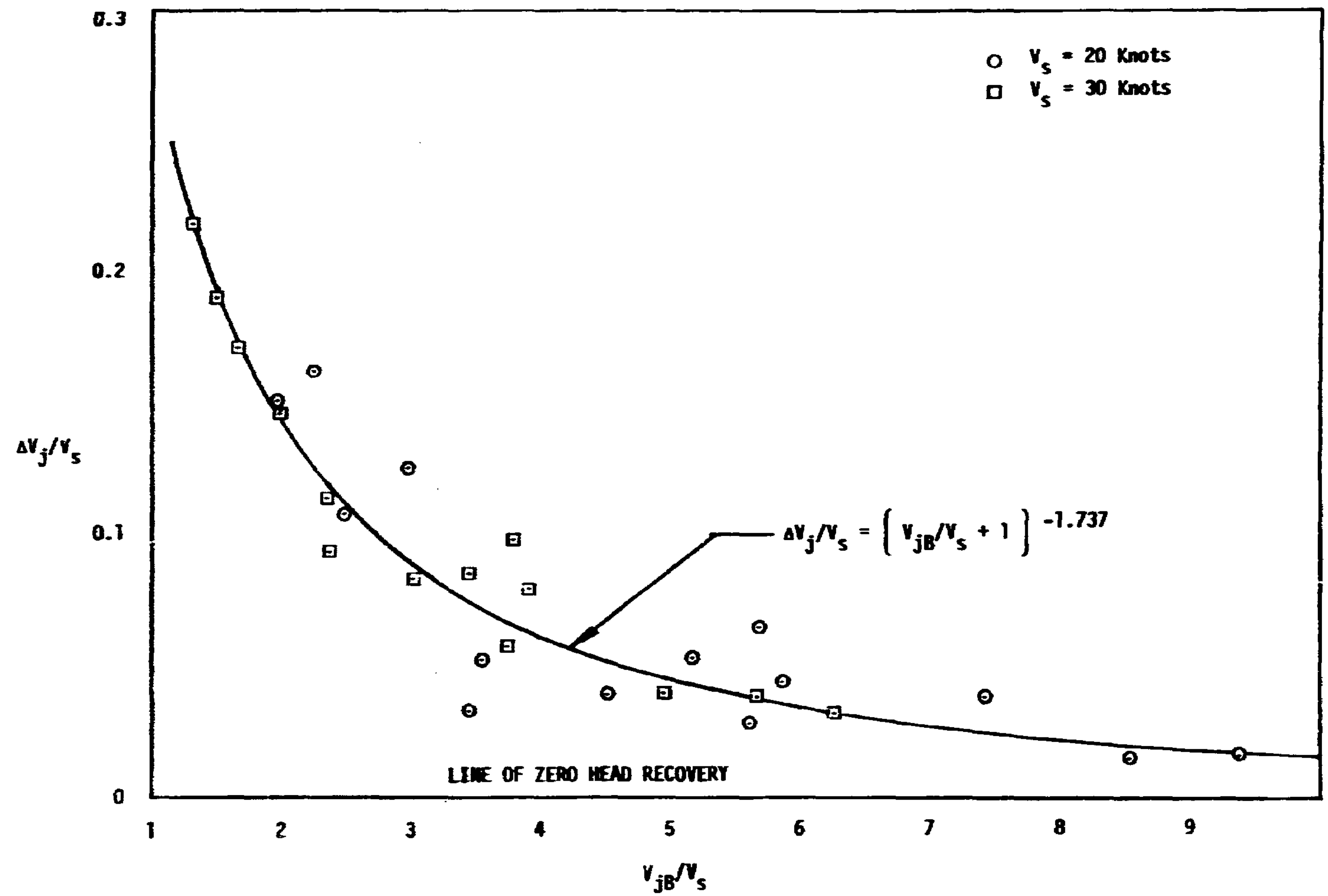

Figure 4 – Differential Jet Velocity Ratio ($\Delta V_j/V_s$) versus Boilard Jet Velocity Ratio (V_{jB}/V_s)

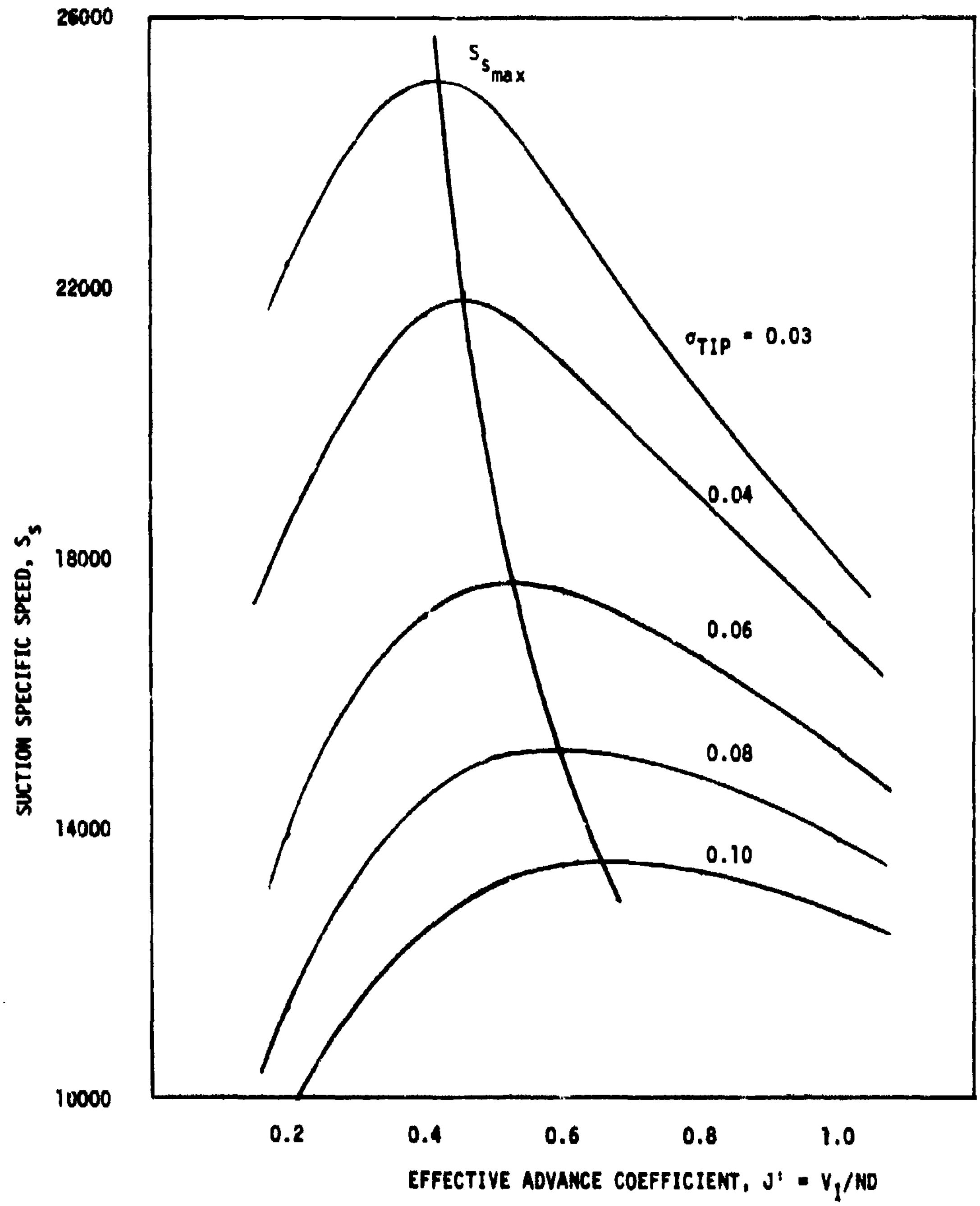

Figure 5 - Suction Specific Speed (S_s) versus Effective Advance Coefficient (J') for Various Impeller Tip Section Cavitation Numbers ($D_h/D = 0.5$)

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